



ASIA **TURBOMACHINERY** & **PUMP** SYMPOSIUM  
12 - 15 MARCH 2018  
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## DRIVETRAIN PROTECTION THROUGH COUPLING DESIGN



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### **ABSTRACT**

This paper is designed to explain types of coupling advanced features and modifications used in turbomachinery, where drivetrain protection is desired. The subject includes a review of flexible coupling types, their characteristics, a detailed outline of their failure modes and two application examples.

### **INTRODUCTION**

Turbomachinery technology is ever evolving and has led to great advancements in every field. Industrial frame and aero-derivate gas turbines have been developed to push the boundaries of power output, efficiency and up-time to meet today's power generation market demands. Gas turbine design optimization has led to machines capable of achieving 50% efficiency; a value unthinkable 20 years ago. Utilizing advanced analytical tools; such as Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA) has resulted in smaller machines, better cooling and notably lower emissions. Electronic control systems, bearing design, dry gas seals and many more developments have led to reduced downtime and increased running periods.

Turbo-compressors have become much more efficient because of today's design advantages. Higher strength, lightweight alloy steels, and tighter machining tolerances enable these machines to operate at very high speed for maximum efficiency. These developments have led to advancements in high performance coupling technology to upkeep even the most stringent of duties and applications. In many cases, the flexible coupling has become an appreciable percentage of the rotor mass and the most influential component in the drivetrain. In addition, compressors are now frequently driven by variable speed synchronous electric motors instead of gas or steam turbines.

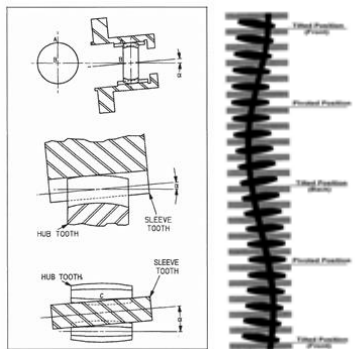
Throughout this paper the term "high performance" or "critical equipment" will be referenced, it is generally recognized as a guideline and can be defined as machines greater than 1000KW rotating at 3000rpm or above where reliability is essential to plant operation and where failure (or maintenance) would result in operational shut down. High performance couplings are defined by the adjoining machines which are expected to operate for prolonged periods of time, typically five years without maintenance.



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## GEAR COUPLINGS

Prior to the advent of flexible membrane couplings, drivetrains utilized gear couplings, the design of these transmit torque and accept misalignment through the meshing and sliding of hardened teeth. (*Figure 1*)



*Figure 1*

Advancements in design, incorporating continuous lubrication to dissipate heat and reduce wear, tighter tolerances and reducing radial clearances helped to minimize potential unbalance but as these couplings wear, their unbalance level changes and eventually must be replaced. Gear couplings also exert a high level of axial force and bending moment onto the adjoining machines, because force is a function of torque for sliding types of couplings.

## FLEXIBLE MEMBRANE COUPLINGS

For the last 40 years, most turbomachinery has used two types of non-lubricated high-performance couplings, disc and diaphragm. Some multinational end users incorrectly interchange these terms so their differences are worth explanation.

### *Disc Couplings*

A disc coupling works by transmitting torque tangentially through a pattern of bolts on a common pitch circle, with the driving bolts transmitting to driven bolts through a series of thin laminations of stainless steel in tension as shown in *Figure 2*. A series of thin plates is used rather than a single thicker section as it is more flexible and poses much lower forces on the connected equipment. Misalignment is accomplished by the bending of the span of material between the driving and driven bolts with the level of misalignment being proportional to the chord length and thickness of the pack. The limit of misalignment comes from the stress limit of bending under tension. Torque capacity is driven by the quantity and size of the bolts. Increasing the number of bolts on the pitch circle increases the torque capacity but reduces the span length between those bolts thus reducing the misalignment capability.

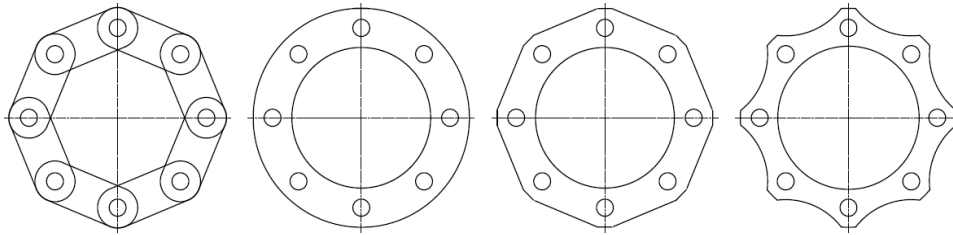


*Figure 2*



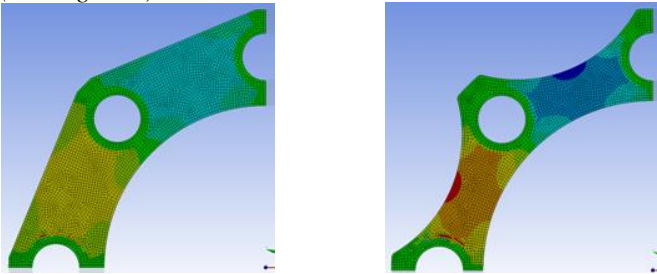
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The origins of the design came from the use of link plates which were connected together via the bolts to form a continuous ring. Some link couplings are still manufactured today for applications with low torque and very large diameter requirements. With the advent of modern manufacturing methods, it became possible to stamp or cut sheets of stainless steel to the required tolerances more cost effectively than producing links. The design of disc profiles varies from manufacturer to manufacturer *Figure 3*. The circular disc is the least preferred of the disc profiles, as it has an uneven torque path resulting in higher stresses than the straight sided or profile blade and is therefore mainly used on industrial pump drives.



*Figure 3*

The preferred straight sided or profiled blades has a more even material distribution throughout the torque path and therefore have a largely even stress distribution. Studying the stress concentration of this section allowed the development of the profiled blade design. (See *Figure 4*)

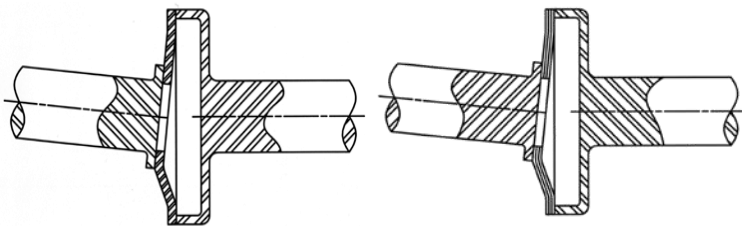


*Figure 4*

Considering the need for the latest drivetrains to be of low inertia and mass to have the least influence upon the adjoining machinery, coupling manufacturers analyze the stresses with advanced Finite Element Analysis (FEA) to optimize this cross-sectional area for torque capacity and misalignment capability, making considerable weight savings. Traditionally, the straight sided blade was considered the low cost option so OEMs accepted higher mass and inertia couplings for better prices, but developments in manufacturing methods have erased this cost difference.

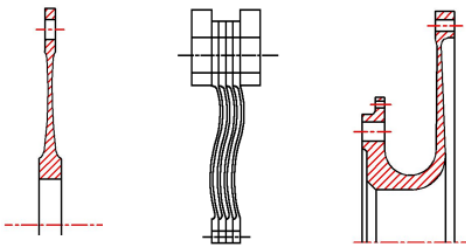
#### *Diaphragm Couplings*

A diaphragm coupling works with either single or multiple plates (diaphragms) transmitting torque in shear from the outside diameter (OD) to Inside diameter (ID), misalignment comes from the diaphragms bending.



*Figure 5*

The profile of the diaphragm is optimized to reduce size, increase flexibility and control stress distribution. The simplest diaphragm profile uses a single plate, machined to create a contour with equal torsional shear stress distribution, while controlling the thickness of the diaphragm to maintain its torque capacity. A thicker diaphragm is able to transmit more torque than a thinner design, whilst a thinner diaphragm has less bend resistance and therefore is softer both axially and in bending. Increasing torque capacity is done by increasing the OD of the diaphragm giving more surface area for torque transmission. Single plate diaphragms are welded to each end of a spacer section to provide parallel misalignment.



*Figure 6 shows the varying diaphragm designs.*

To avoid the need for welding of the diaphragm to the spacer, a single contoured diaphragm that fastens to the spacer was introduced into the market in the mid 90's. The shape of this flex-element can be designed from an "I" shape (shown in *Figure 6*) to a "U" shape depending on torque and axial deflection requirements. The other widely used design is the multiple convoluted diaphragm which incorporates the benefits of the single plate designs and the design philosophy of the disc coupling in that it uses multiple plates of thinner stainless steel. With multiple thin section diaphragms, the bending and axial stiffness is greatly reduced and the multiple plates can transmit greater levels of torque for a given diameter. The pack is designed so that in most cases only 50% of the diaphragms are needed to transmit the application duty torque. This inherent redundancy means fatigue failure is gradual and often be detected by train vibration monitoring. Both the "I" shaped and multiple convoluted diaphragm type offer a field interchangeable flex-element that some end users consider an advantage when the cost of spares is a consideration.



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## COUPLING SELECTION AND DESIGN

The standard for turbo-compressor couplings used in refining is API-671/ISO10441. The scope of this standard defines the design and selection of couplings for these trains and is often considered the “Bible” for all centrifugal gas compression applications. To properly select the coupling, the manufacturer requires key information to be provided to them. As a guide for this, API 671 includes a data sheet (Figure 7) which should be completed by the purchaser, to ensure that the coupling supplier has the necessary information to design a coupling for the application, and the coupling supplier to list data and ratings for the coupling. There is also a section for adding accessories such as hydraulic tooling (and gauges) along with any additional specifications that may be applicable. This should not be considered the be-all and end-all of information required but rather a basic guideline.

COUPLING DATA SHEET		Page _____ of _____	
API 671-4th Edition/ISO 10441-2007		Job No. _____	
By _____		Purchase order No. _____	
By _____		Date _____	
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By _____		By _____	
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2. Applicable to: <input type="checkbox"/> Proposal <input type="checkbox"/> Purchase <input type="checkbox"/> As-built			
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Figure 7

One key point for consideration is the inclusion of a service factor, API 671 4<sup>th</sup> Edition section 6.7 stipulates:

*“Unless otherwise specified, the coupling shall have, as a minimum, a service factor of 1.5 for a metallic flexible-element coupling. The service factor may be reduced if mutually agreed upon and if the characteristics of the driver and driven machines are well understood or all reasonable attempts to achieve the specified torque service factor fail to result in a coupling mass and subsequent overhung moment commensurate with rotor dynamic requirements (ref. API-617). In no case shall the service factor be reduced to a value lower than 1.2 based on normal operating point. “*

The terms service factor and safety factor are very commonly misunderstood, warranting further explanation. Service factors are applied to the steady state torque to allow for site specific conditions as well as equipment and other variations resulting in higher than expected torque. Safety factors are used to cover uncertainties in a coupling design such as material properties, manufacturing tolerances and analytical assumptions in stress analysis.

The rating of a flexible element coupling is calculated based upon combined stresses. Gas or steam turbines, centrifugal compressors and pumps have a smooth torque output and therefore are considered as having a steady state torque. Axial deflection is also considered steady state whilst angular misalignment gives an alternating stress. It is the combination of these stresses that determine the coupling’s torque rating. Most couplings are rated at full torque, 100% angular and axial deflection simultaneously. Flexible membrane couplings are often pre-stretched to accommodate axial thermal growth of the machines. Because of today’s laser alignment tools, angular misalignment and parallel offset are achieved at a fraction of the rated 0.20 degrees. Although API-671 requires a minimum safety factor of 1.25, high performance couplings rarely fail as they don’t normally operate anywhere near their limits.



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API-671 stipulates that “The vendor shall state the continuous torque rating at the coupling rated speed, when simultaneously subjected to the maximum continuous angular misalignment and the maximum continuous axial displacement. The fatigue factor of safety at the continuous torque rating and any published combination of speed, angular misalignment and axial displacement shall be determined using the proportional increase method with either the modified Goodman diagram or constant-life curves, together with the mean and cyclic stresses induced in the flexible element under the evaluated conditions. If the modified Goodman diagram is used, the fatigue factor of safety shall be not less than 1.25. If the constant-life curve is used, the fatigue factor of safety shall be not less than 1.35. “

Figure 8 shows the makeup of a modified goodman diagram, with the API defined Safety Factor added. Coupling manufactures can calculate the resultant stresses from torque, axial deflection and angular misalignment and plot them to ensure the coupling is suitable for the lifetime of the machinery as can be seen in the example in Figure 9.

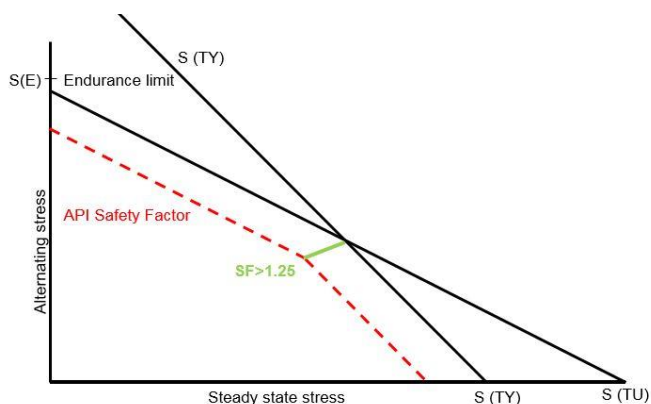


Figure 8

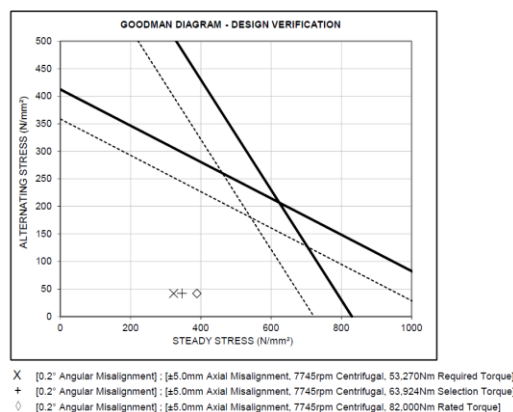


Figure 9

The use of such tools allows the designer to determine the weakest point of the coupling. It is widely expected (and specified by some end users and original equipment manufacturers) that couplings be designed so that the weakest point of the coupling is the flexible element. There are situations, such as torsionally tuned couplings, where the torsion bar spacer maybe the weakest point and should be highlighted for review. In practice, the ability to accurately predict the precise failure point of a flexible element is more uncertain and therefore should not be considered as a “fuse” in the train. There are more accurate methods for torque release or backup available, if required for a given application.

As mentioned earlier, because of the growing popularity of motor driven compressors, new and modernized plants have seen a rapid growth of synchronous and VFD electric motor compressor trains. Most, if not all coupling manufacturers have taken their turn with catastrophic failures due to unpredicted transient torsional peak torque at start-up (Figure 10). The amplitude of such peak torques can be more than 10 times duty torque, very difficult to determine and may require a more detailed analysis. API-671 addresses both of these conditions as follows:

For motor driven applications, API 671 4<sup>th</sup> Edition section 7.3 further stipulates:

“For applications where cyclic torques can occur (such as from synchronous motor, generator or reciprocating compressor), the coupling design shall be determined by completing a fatigue analysis. The analysis shall take into account the normal operating conditions in conjunction with the transient conditions. Depending on whether the cyclic loads are considered for a limited number of occurrences or considered to be infinite, either a low-cycle fatigue analysis or high-cycle fatigue analysis shall be performed for the various torque-transmitting components in the coupling. For low-cyclic fatigue analysis, the resultant mean and cyclic stresses plotted on a fatigue diagram shall fall under the applicable life curve (i.e.  $10^4$ ,  $10^5$ ,  $10^6$  or  $10^7$  cycles).”





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VFD electric motor driven compressor trains have become popular because of their many advantages. Compared to steam turbines, they require little maintenance, have a higher efficiency and complete control of load torque across the entire speed range. In addition, these drives typically eliminate the need for a speed increasing gearbox. There can be problems, however, associated with running trains at various speeds. Certain speeds must be avoided to prevent operating on or near torsional critical frequencies. The problem is that in many applications, the actual frequency and amplitude of these critical speeds are unknown or don't fall within the calculated range. This issue has opened a new market for coupling torque meter systems. Modifying the coupling to incorporate a torque meter (See Figure 11 and Figure 12), can identify and record the torque frequency and amplitude, to enable adjustment avoiding operating in these speed ranges. There are two solutions for end users to consider; the first is an inexpensive, "strain gauge" system (Figure 11), which is accurate but may have limited useful life, therefore is generally used for commissioning only. The second, a more robust, all metallic, "phase shift" type design (Figure 12), can be used by users who want to monitor torque output constantly, in addition to identifying torsional frequencies.

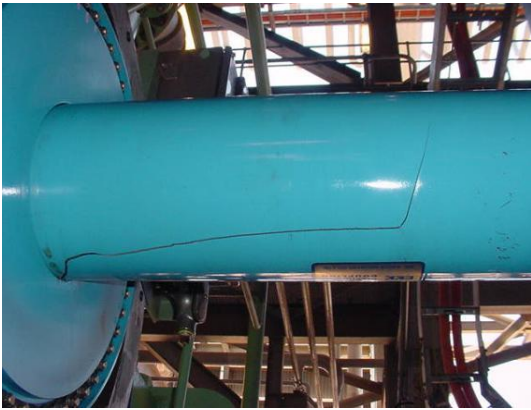


Figure 10

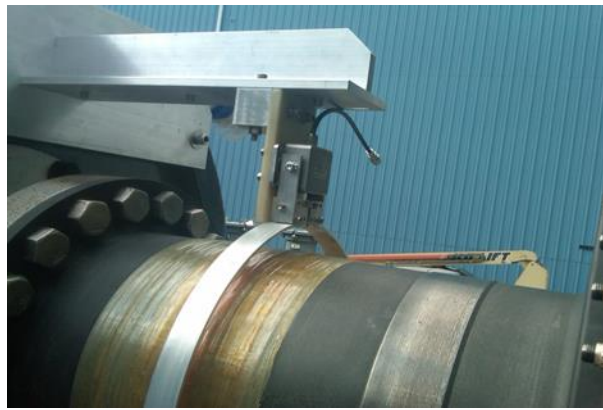


Figure 11

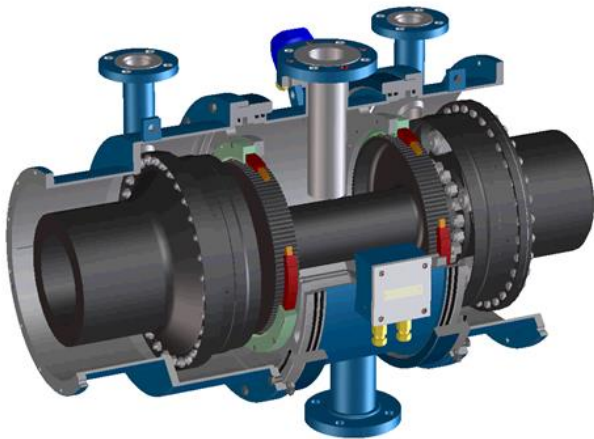


Figure 12



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## FAILURE MODE EFFECT ANALYSIS

The following section outlines the causes of failures and failure modes.

### *Materials and Corrosion*

All high performance membrane couplings are subject to Stress Corrosion Cracking (SCC), which is the combined action of static stress and corrosion, leading to cracking or embrittlement of metal in corrosive environments. Austenitic stainless steels exhibit a high susceptibility to stress corrosion cracking when the temperature exceeds 65° C (*reference Handbook of Stainless Steels*). Diaphragms made from alloy steel require coating for protection. Diaphragms made from 15-5 PH or 17-4 PH stainless (H1025) are inherently highly resistant to stress corrosion cracking at sustained stress levels much greater than those seen under normal running conditions and therefore are acceptable for H<sub>2</sub>S and chlorine environments without coatings. Machined diaphragms are often shot peened to reduce residual stresses imparted during manufacture, removing any surface imperfections. This process also provides a layer of compressive stress, and stress corrosion cracking cannot occur in an area of compressive stress. Disc couplings are typically made from 300 series stainless, therefore in a corrosive environment may require changing to Inconel or a coating applied

### *Disc Coupling Failure*

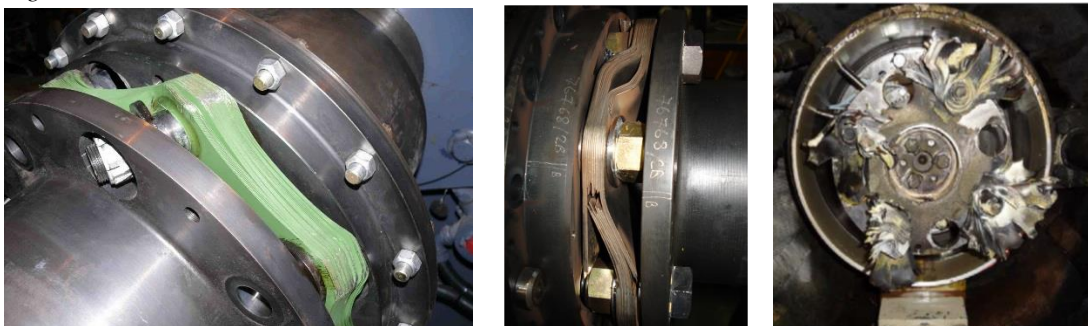
There are 3 main failure modes for the flexible membranes of a disc coupling.

1, Excessive misalignment causes fatigue failure around the washer with the outside lamination failing first and then prorogating through the pack of laminations in direct contact as seen in *Figure 13*. Depending on the level of misalignment seen, this propagation through the membrane pack is generally gradual with torque being carried through the remaining membranes. Couplings that operate at very high misalignment can cause rapid failure. Vibration monitoring in the adjoining machines can detect the failure before becoming catastrophic.



*Figure 13*

2, An overload in torque generally collapses the compressive leg of the membrane pack between the driven and driving bolt as seen in *Figure 14*.



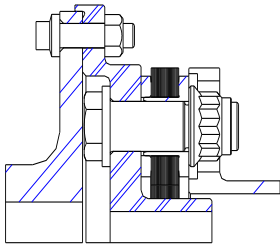
*Figure 14*





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A large, single torque overload will cause a sudden failure of membranes this failure can cause the spacer to flail and be ejected from the machine. To ensure this section is controlled, an anti-flail device (*Figure 15*) should be incorporated into the design (a mandatory requirement under API 671). Designs in which the spacer is controlled by the coupling bolts only should be avoided.



*Figure 15*

Note. A coupling guard is not designed to withstand the kinetic energy of a flailing or ejected spacer and should not be relied upon as a primary safety device. Its purpose is to deflect foreign articles from ingress and personnel protection only.

3, Disc couplings can fail due to fretting of the membrane laminations. Under misalignment the membranes bend with the outside membranes bending furthest. The action of this bending causes the surfaces to slide against one another. Over time, this micro-movement wears the surface of the materials and can lead to fretting corrosion failure. Manufacturers try to overcome this with different methods. A large preload achieved by using a larger bolt can give sufficient clamp load to avoid movement at the expense of increased size and mass. The other method is to apply a low friction coating, similar to Teflon, that allows micro movement without causing fretting failures. *Figure 16* shows fatigue test samples with and without coating, run under identical high misalignment conditions.



*Figure 16*

#### *Diaphragm Coupling Failure*

Diaphragm coupling failure mode is quite different to a disc coupling.

Failure of a single plate diaphragm through misalignment can be seen as a ripping of the plate below the connection PCD, as seen in *Figure 17*. Failure through axial misalignment can be seen as a clean perforation below the connection PCD as seen in *Figure 18*. In either case, unlike a disc coupling that continues to drive through the bolts after a catastrophic failure, diaphragm couplings fail in torsion and disconnect the driving and driven equipment.



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Figure 17



Figure 18

Multiple plate diaphragm couplings fail from the highest stress concentration which is the plate ID and propagate outwards. Unlike a disc coupling in which all laminations are in contact, the inbuilt gap between diaphragms can arrest the propagation. *Figure 19* shows the outer diaphragm of a failed pack. *Figure 20* shows the second diaphragm of the same pack after months of operation. There was no damage to it or any of the other diaphragms in the pack.

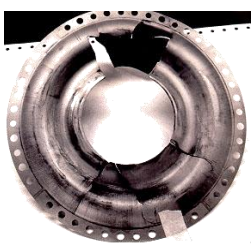


Figure 19

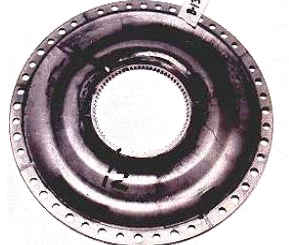


Figure 20

Torque overload of a diaphragm generally happens in shear at the diaphragm ID see *Figure 21* and *Figure 22* and leads to total disconnection of all drive, unlike a disc coupling in which the coupling bolts act in shear and may continue to rotate the coupling for an indeterminate period of time. This can cause damage to the adjoining machines and local scenery, due to parts flailing at high velocity and uneven drive load. It is impossible to determine the fatigue failure point of diaphragm couplings because stresses from actual alignment, compared to the rated usually differ, so the coupling should not be designed to act as a “fuse”.



Figure 21



Figure 22

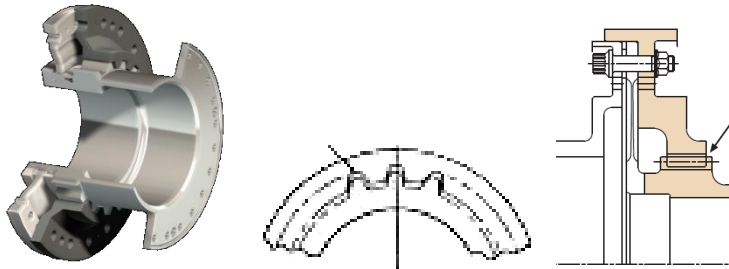
Some multi-national oil and gas end users have specifications which stipulate that in the event of a total membrane failure the adjoining machines should be brought to an immediate stop, and prohibit the use of designs which continue to drive through bolts.



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## BACKUP DRIVES

To mitigate the potential for disconnection of drive in drivetrains where continuous drive is required, a backup system can be implemented within the coupling of various designs. Both disc and diaphragm couplings can incorporate a dental backup drive, consisting of a set of gear teeth which have enough clearance during normal operation to avoid contact. In the event of a disc or diaphragm failure the clearance is taken up and the teeth used to provide drive as shown in *Figure 23*.

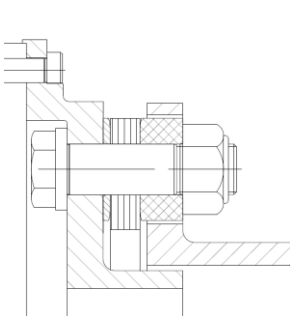


*Figure 23*

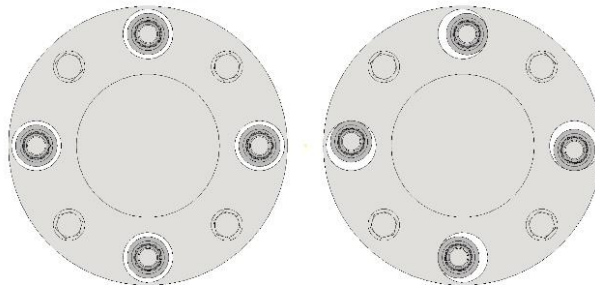
Unlike a traditional gear coupling, the teeth have no form of lubrication to reduce friction and heat, therefore would be expected to wear quickly. Although this design has been utilized in many couplings seen in service, no records of failure using such a device have been published, meaning very little indication of running time can be given.

There are several disc pack coupling modifications designed to protect from catastrophic failure.

The simplest design incorporates the addition of an overload collar (often made of a “spark resistant” material, such as bronze or a bronze alloy) under the coupling bolt *Figure 24*, in the event of a membrane failure the overload collar acts as a bush within the clearance hole to provide drive *Figure 25*.



*Figure 24*



*Figure 25*



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The use of overload collars is widely seen within process pumps where motor characteristics are less well known or controlled, as such many examples of their use can be identified. *Figure 26* is from a coupling which suffered a large torque overload, the overload collar gave temporary drive whilst the machines coasted down. As with the dental backup design the overload collar has no form of lubrication, therefore the significant heat created has melted the overload collar and associated fasteners. *Figure 27* comes from a bearing failure, the overload collar provided drive until the heat produced melted the spacer material around the collar. The design of coupling was not controlled under API 671 and did not have an anti-flail device, the containment of the spacer piece being controlled only by the overload washers. With the collapsing of the material around the overload washer it is clear that the spacer could have been ejected if rotation had continued any longer.



*Figure 26*

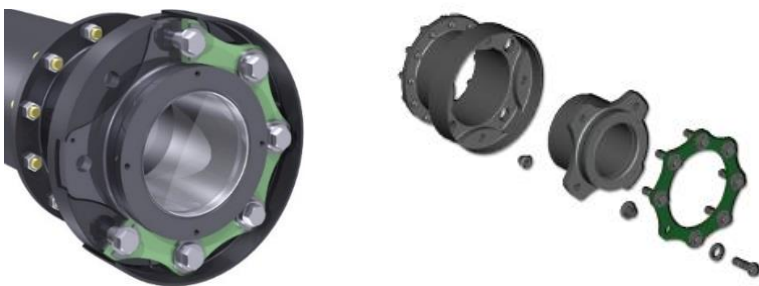


*Figure 27*



It should be noted, that there are no “spark free” materials in the turbomachinery coupling world. Even if the flexible membrane could be designed to eliminate sparking in the event of a failure, the other components of the coupling and the machines they connect to are typically made of steel, therefore it is impossible to be classified as “spark-free”.

A more robust design, commonly specified to prevent runaway of steam turbine drives in the event of membrane failure, is commonly referred to as a co-planar design. This design features interlocking driving and driven flanges that engage only in the event of a catastrophic membrane failure. In this failure mode, the driving hub engages with the mating sleeve to allow the connected equipment to be shut down without a catastrophic coupling failure. This back-up device is only intended to allow the user to shut down the train. Clearances are designed into the interlocking flange flanks so they will not come in contact during normal operation, even at maximum rated misalignment, see *Figure 28*. Another benefit of this design is that even though the coupling mass is greater, the overhung moment of the ½ coupling is often lower because its centre of gravity is further up the rotor shaft and therefore closer to the bearing. For high speed compressors where the 2<sup>nd</sup> lateral critical is a concern, this design may provide an acceptable failure mode for a disc coupling whilst offering a solution to lateral rotor dynamic issues.

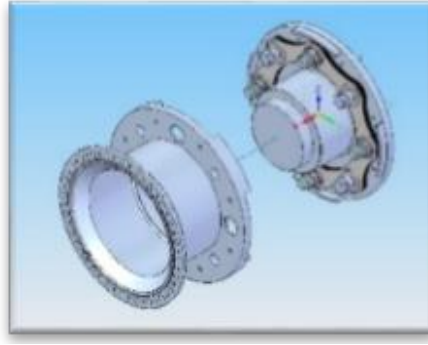


*Figure 28*



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A less popular, but sound design for low torque applications, adds castellated shrouds to the flanges either side of the flexible elements. The castellated shrouds are orientated so that they interlock within each other. In the event of flexible element failure, the castellations provide temporary drive see *Figure 29*.



*Figure 29*

With all of these methods it should be reiterated that these mechanisms have no form of lubrication to prevent wear, and as such could fail very quickly. Careful consideration to their attributes should be given to ensure suitability within the drivetrain design.





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## OVERLOAD PROTECTION

Within the coupling, it is possible to incorporate a number of devices to overcome the potential damage caused by a torque overload. The most common is to incorporate a mechanical shear device, either as a single spool piece, commonly referred to as a shear spacer *Figure 30* or shear pins which have a reduced section groove on a common pitch circle, *Figure 33*. In the event of a torque overload, the reduced section groove(s) will shear, disconnecting the turbine from the driven equipment.

The design of such devices needs to be tightly controlled to achieve an accurate disconnection. A device which shears below the specified torque would lead to premature disconnection causing downtime of the system, whilst high torque disconnection could cause damage to the prime mover at huge expense. To avoid failure through fatigue it is paramount to have a separation factor of 2.5 to 3 between normal running conditions and shear point of the overload device.

Looking at both designs in more detail.

### Shear Spacers

A shear spacer design consists of a single spool piece with reduced section groove of a calculated diameter determined to shear at a given torque. In order to maintain alignment, stop flailing of the two sheared sections and allow run-down in a controlled manner, a bearing is included. The design of this run-down bearing can differ in design with either a plain bearing manufactured from a brass material (*Figure 30*) or dynamic ball or roller bearings being used.

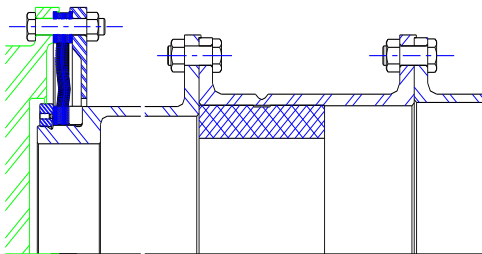


Figure 30

The introduction of the bearing arrangement has to be controlled to avoid influence upon the shear disconnection point. A design with heavy interference between a bearing or liner housed within the tube can increase the shear point.

During testing of a shear spacer by Ameridrives Ltd *Figure 31*, the results matched that of a “typical” tensile test *Figure 32*, showing no influence from the plain bearing used. Careful design and manufacture of the shear groove is needed, the groove often being shot peened after manufacture to reduce the possibility of failure through fatigue, giving a localized compressive stress and removing surface imperfections.

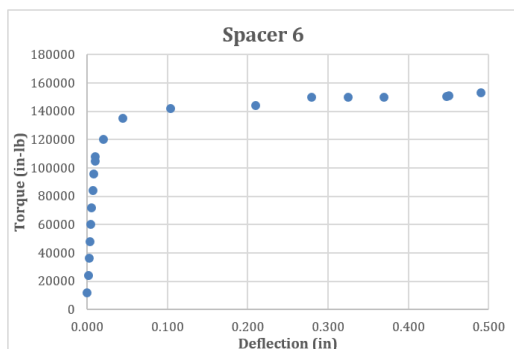


Figure 31

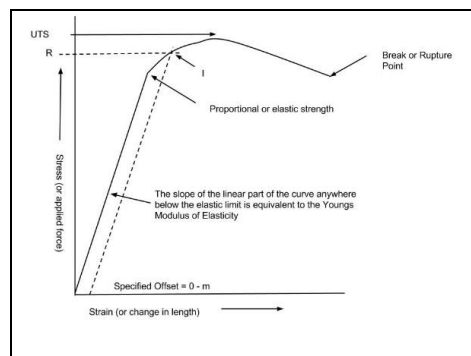


Figure 32



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### Shear Pins

A shear pin arrangement consists of a series of pins located in 2 flanges, the pins having a calculated groove diameter, positioned on a common pitch circle *Figure 33*. A bearing is included to control rotation after disconnection.

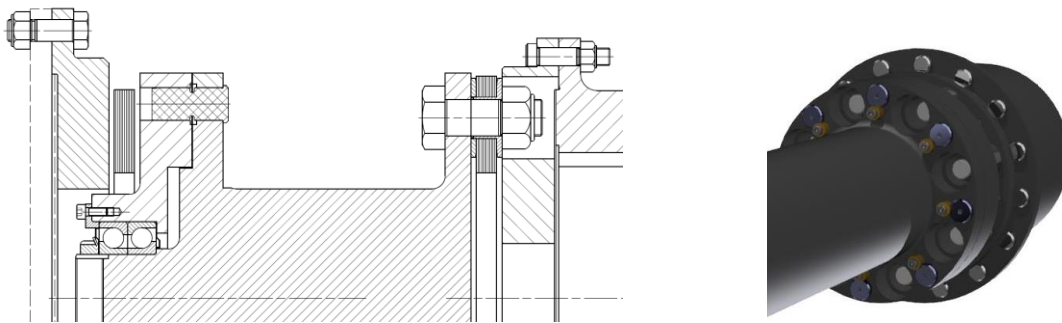


Figure 33

To ensure disconnection accuracy, the precise shear of multiple pins concurrently is required. This relies upon the true position of the pin holes in both flanges to ensure all pins have equal torsional loading. A design incorporating 4 pins with 0.1mm (0.004 inch) true position, results in contact loading of 85%. Increasing the tolerance on hole position to 0.25mm (0.010 inch), sees this reduce to 75%, thus reducing the accuracy of disconnection. The rigidity of the shear pin arrangement is resultant upon the fit of the shear pins within the holes and the robustness of the bearing arrangement. A flexible joint can cause bending fatigue upon the shear pins and an increased face run out; imposing axial thrust loads upon attached machinery.

Whilst the shear pins are replaced after any disconnection without the need for the coupling removal the bearing arrangement is generally of a “fit and forget” design: meaning that the design has to be capable of multiple running periods, whilst also enduring the effects of long periods of non-use. For any dynamic bearing, extended periods of non-use can do significantly more damage than constant running, as rollers or balls can cause indentations into the surface of cages and races often referred to as Brinelling. This damage can cause increased wear and even bearing failure. Shear pin retention is done in 2 ways; either a radial “grub” screw acting upon a groove within the shank of the pin, or an axial retention plate is used.

With reference to the above it is possible to see the advantages and disadvantages of both designs

#### Shear Spacer

- ✓ Low initial cost
- ✓ Not subject to same bending fatigue of pins
- ✓ Simple design
- ✗ Increased downtime in replacing spool section
- ✗ Costly to replace after failure and high inventory cost to stock
- ✗ Susceptible to fatigue from reversing torques
- ✗ Generally require longer space envelope than a shear pin design

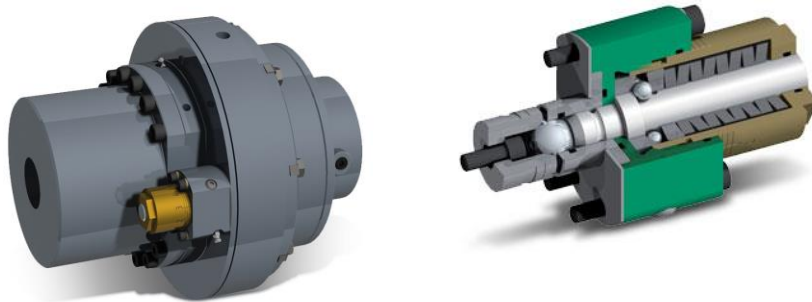
#### Shear Pins

- ✓ Low cost replacement pins
- ✓ Reduced downtime - quick turnaround replacement of pins
- ✓ Suitable for short couplings
- ✓ High fatigue applications can be “lived” for preventative pin change
- ✓ Reasonably easy adjustability with different groove diameter pins
- ✗ Expensive initial design, incorporating bearings, additional shear pin flange, precision bored holes and manufactured pins
- ✗ Subject to bending fatigue
- ✗ Life can be difficult to predict in applications without adequate ratio between shear value & normal torque
- ✗ Temptation to increase shear torque due to failures, risking equipment protection

#### *Other Protection Methods*

Other methods of protection can be introduced to prevent overload torque damage.

Safety element torque limiting devices which incorporate an array of modules on a common pitch circle can be added *Figure 34*. Each module containing a spring loaded ball acting upon a hardened detent pocket gives easy resetting after an overload, however due to their size, they are better suited to low speed applications.



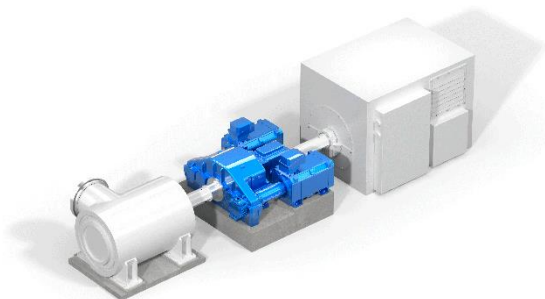
*Figure 34*

A hydraulic torque limiting device which uses hydraulic pressure to drive torque in friction can be incorporated into the flexible coupling. During overload a shear tube shearing releases pressure and cutting drive (*Figure 35*). Drive can be reconnected by replacement of shear tube and repressurization.



*Figure 35*

The traditional mechanical reduction gearbox can be replaced with a hydrodynamic gearbox which dampens any torque overload from passing through the drivetrain (*Figure 36*).



*Figure 36*



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## EXAMPLE APPLICATIONS

### Power Generation

In the field of power generation, the largest market for flexible couplings is distributed power, gas turbine driven generators (commonly referred to as gen-set's). The use of these packages (*Figure 37*) often necessitating a gearbox because the power turbine speed does not match the generator output speed whether 50Hz or 60Hz. To enable the power generated to be added to a power grid, the power output needs to be synchronized. If the power output doesn't match the grid being joined *Figure 38*, (commonly referred to as a "malsynchronization" or "malsync"), the inertia of the grid being larger will cause the generator's rotor to rapidly increase or decrease speed to match. This speed change is seen as a spike or overload in torque.

These torque fluctuations can be calculated by the generator manufacturer to analyze the potential for damage within the system. A small torque overload maybe absorbed within the system, whilst a large overload could cause damage to the various components of the drivetrain. The integration of an overload device into the coupling ensures that regardless of the severity of torque fluctuation, the turbine is protected.



Figure 37

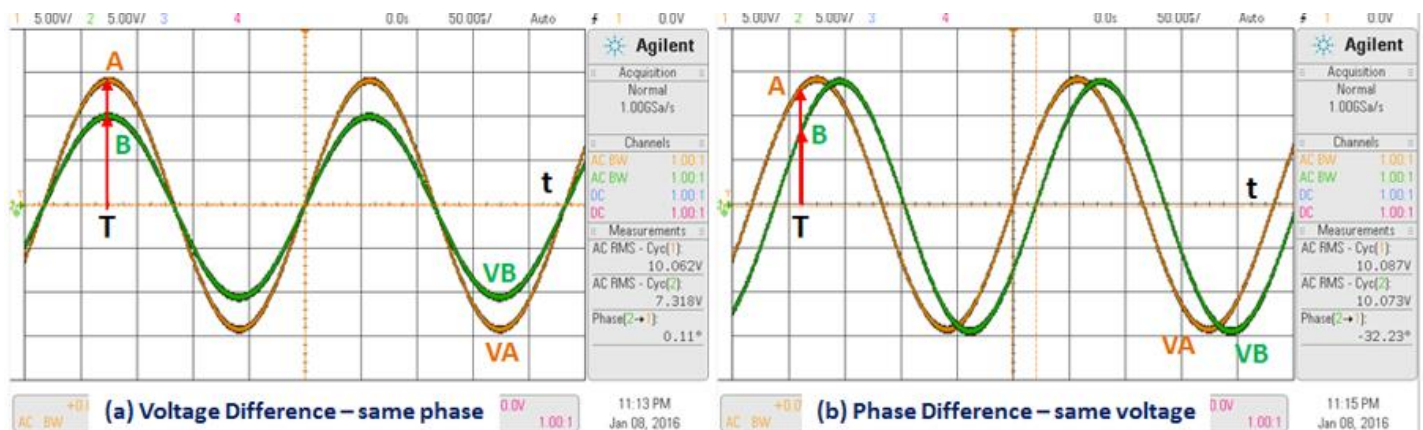


Figure 38



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### *Plastic Processing*

The production of plastics and polymers is a huge part of the turbomachinery market. Production of olefin products require their own special requirements and many books and papers have been written about these. This paper addresses methods of modifying the flexible coupling to protect the critical equipment. A good example of this is a polymer pump. Manufacturing plastic resin starts with various hydrocarbons being chemically processed in large, continuous-flow polymerization plants. The polymer resins are then further processed and the mixed resin is cooled to form pellets, chips or beads. These materials are used to manufacture countless finished products through a variety of processes, including extrusion, injection, blow and rotational molding. Large-capacity motor-driven pumps are the heart of the polymer processing plants. Durable couplings are required to connect the pump rotors to gearboxes. Coupling failure would result in the mixing system coming to a halt and allowing the molten plastic to cool and set in and around critical equipment. In older plants rotary screw mixing pumps were driven by tandem gear couplings (*Figure 39*), and although the application is low speed, the disadvantages of gear couplings previously outlined, has seen them being replaced with diaphragm couplings (*Figure 40*), that can operate without maintenance for years rather than months.



*Figure 39*



*Figure 40*





## CONCLUSION

A flexible membrane coupling can be considered as more than a simple link between machinery. Understanding the various features and benefits of high performance couplings is important, but for lead specifiers and users, it is equally important to be familiar with modifications that can be made to this critical component to make it capable of preventing failure, warning operators of a potential or imminent failure and protecting the connected equipment from catastrophic damage in the unlikely event of such failure. Though often unknown, these relatively simple coupling modifications can save millions of dollars in downtime and repair costs for critical turbomachinery.

## REFERENCES

*American Petroleum Institute, Standard 671. Special-purpose Couplings for Petroleum, Chemical and Gas Industry Services, 4<sup>th</sup> Edition, 2007.*

*J.P. Corcoran, C Wolford, 2016, Couplings – Balancing Tutorial & “New Developments in Gas Turbine Couplings”, Proceedings of Asia Turbomachinery & Pump Symposium, Turbomachinery Laboratory, Texas A & M University, College Station, Texas*

## ACKNOWLEDGEMENTS

The author would like to thank the Turbomachinery and Pump Symposia advisory committees for the opportunity to present this paper. Thanks especially to Turbomachinery Symposium advisory committee member Mr Christian Wolford for reviewing the drafts of the tutorial and providing excellent feedback.